REPORT DOCUMENTATION PAGE	Form Approved OMB No. 074-0188

1. AGENCY USE ONLY (Leave blank)

2. REPORT DATE

3. REPORT TYPE AND DATES COVERED

4. TITLE AND SUBTITLE
Evaporative Compressor Cooling For NOx Suppression and Enhanced Engine Performance for Naval Gas Turbine Propulsion Plants

5. FUNDING NUMBERS
N/A

6. AUTHOR(S)
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8. PERFORMING ORGANIZATION REPORT NUMBER
N/A

9. SPONSORING / MONITORING AGENCY NAME(S) AND ADDRESS(ES)

SERDP
901 North Stuart St. Suite 303
Arlington, VA 22203

10. SPONSORING / MONITORING AGENCY REPORT NUMBER
N/A

11. SUPPLEMENTARY NOTES

12a. DISTRIBUTION / AVAILABILITY STATEMENT
Approved for public release: distribution is unlimited.

12b. DISTRIBUTION CODE
A

13. ABSTRACT *(Maximum 200 Words)*
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14. SUBJECT TERMS
water fog injected; WFI; gas turbine cycle; SERDP; SERDP Collection

15. NUMBER OF PAGES
10

16. PRICE CODE N/A

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
unclass.	unclass.	unclass.	UL

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std. Z39-18
298-102

CP-042

EVAPORATIVE COMPRESSOR COOLING FOR NOx SUPPRESSION AND ENHANCED ENGINE PERFORMANCE FOR NAVAL GAS TURBINE PROPULSION PLANTS

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ABSTRACT

Water, in the liquid or vapor phase, injected at various locations into the gas turbine cycle has frequently been employed to improve engine performance while simultaneously reducing NOx emissions. Commercial steam injected gas turbines have been designed to inject small amounts of steam (less than 15% of air flow), generated in a heat recovery boiler, into or downstream of the combustor. Recently, it has been proposed to inject larger amounts of water (as high as 50% of air flow) and operate combustors near stoichiometric conditions. All these methods increase turbine mass flow rate without increasing air flow rate and consequently increase specific power. The increase in specific power for naval applications means smaller intake and exhaust stacks and therefore less impact on topside space.

The present paper presents a new concept, in naval propulsion plants, to decrease NOx production and increase specific power with a water fog (droplet spray) injected (WFI) directly into the inlet of the engine compressor. The simulated performance of a simple-cycle gas turbine engine using WFI is reported. The paper describes the computer model developed to predict compressor performance resulting from the evaporation of water passing through the stages of an axial flow compressor. The resulting effects are similar to those of an intercooled compressor, without the complications due to the addition of piping, heat exchangers, and the requirement for a dual spool compressor. The effects of evaporative cooling on compressor characteristics are presented. These results include compressor maps modified for various water flow rates as well as estimates of the reductions in compression work and compressor discharge temperature. These modified compressor performance characteristics are used in the engine simulation to predict how a WFI engine would perform under various water injection flow rates. Estimates of increased output power and decreased air flow rates are presented.

NOMENCLATURE

A	surface area of a droplet, ft^2
C_p	specific heat, $Btu/(lb_m R)$
D	droplet diameter, ft
$g_{m,1}$	mass transfer conductance, lb_m/ft^2 sec
h_{fg}	enthalpy of evaporation, Btu/lb_m
h_c	convective heat transfer coefficient, $Btu/(ft^2$ sec R)
$j_{1,s}$	mass flux at liquid surface, lb_m/ft^2 sec
m_1	mass fraction of vapor
Re	Reynolds No.
Sc	Schmidt No.
Sh	Sherwood No.
St	Stanton No.
Pr	Prandtl No.
t	time, sec
T	temperature, R
V	velocity, ft/sec
$\delta_{1,2}$	diffusion coefficient of vapor in air, ft^2/sec
ν	kinematic viscosity, ft^2/sec
ρ	density, lb_m/ft^3

Subscripts 1 and 2 represent the vapor and air respectively
Subscript s represents surface properties
Subscript e represents bulk properties
Subscript l represents liquid properties

INTRODUCTION

Gas turbine engines have been selected for use in modern naval vessels for several reasons. Most notable of these are the high power to weight ratio and the rapid startup and response times of the gas turbine engine. One detrimental factor of the gas turbine engine for naval use is the large volumes of intake and exhaust ducts necessary to bring in, and remove, the large quantities of air required by the gas turbine engine. In order to minimize the intake of salt-water spray, the air intakes are usually located high in the ship's superstructure. This space in the superstructure is valuable for weapons, electronics, and other mission-critical equipment. It would, therefore, be beneficial for naval ship installations to increase the engine specific power, that is, the engine output power per unit mass flow of air through the engine. The usual units for specific power are shaft horsepower-second per pound of air (hp-sec/lbm.)

It has long been recognized that water, in the liquid or vapor phase, injected at various locations into the gas turbine cycle can be employed to improve engine performance while simultaneously reducing NOx emissions. Water added in the combustor of a gas turbine engine has the dual effect of (1) lowering the combustion temperature, and thereby reducing the generation of NOx, and (2) increasing the mass flow rate through the turbine sections and thereby increasing engine power. This addition of water to the combustor, however, results in the increase of engine specific fuel consumption, that is, mass of fuel per horsepower-hour (lbm/hp-hr.)

Steam augmented gas turbine engine cycles have been proposed (Ediss, 1970, Cheng, 1978, Brown, and Cohen, 1980, Urbach, 1985, 1993, 1997a, and Rice, 1993) to potentially increase power density, and decrease NOx emissions. Most of these cycles utilize a heat recovery boiler to produce steam that is introduced into the engine at either the combustor or turbine section, or both. These engines, because they recover energy from the exhaust gases to operate the heat recovery steam generator, have the additional benefit of increased fuel efficiency.

This paper considers the possible benefits from the injection of a water fog (droplet spray) directly into the inlet of the engine compressor (Water Fog Injection.) The WFI concept has been considered as early as 1950, when Wilcox and Trout (1950) suggested that compressor work and discharge temperature could be reduced by injecting a liquid into the gas to be compressed. More recently, Urbach, et al., (1997b) reported the results of a thermodynamic study of the effects of water fog injection on a naval propulsion gas turbine engine. Two experimental studies have produced preliminary data. Zheng Qun, et.al., (1997) have reported the effects of water spray into the centrifugal compressor of a S1A-02 gas turbine engine. Stambler (1997) reports a 15.5% increase in power output from a Fr7E peaking plant when the compressor is cooled with an evaporative water spray.

EVAPORATIVE COMPRESSOR COOLING

Water fog injection into the inlet of a naval propulsion gas turbine engine was envisioned to be a low cost option for reducing NOx emissions as well as increasing engine specific power. Experimental work conducted by John Emory and Jack McGoarty of the Navy's land-based LM2500 engine facility at Philadelphia (reported by Urbach, et al., (1997b)) showed (see Fig. 1) that the addition of 4.45 gpm of water to the inlet of the engine compressor reduced NOx emissions by approximately 33 ppm over the engine operating range. Water fog injection into the compressor inlet also acts to cool the inlet air as well as the air passing through the compressor stages until all the water is evaporated. This cooling yields effects similar to those obtained from an intercooled compressor. The thermodynamic study by Urbach, et al., (1997b) reported increases in power output of nearly 34.5% when injecting 24 gpm of water into an engine air flow rate of 134 lb_m/sec, as shown in Fig. 2, while simultaneously providing a reduction of 8.8 % in specific fuel consumption. Additional results show that for a nominal 25,000 shaft horsepower engine, reductions in compressor discharge temperature of 200 degrees or more could be obtained (see Fig. 3.) If full advantage of this reduction in compressor discharge temperature were taken by the addition of a recuperator, significant improvement in thermal efficiency could be obtained (see Fig. 4.) These favorable results obtained from the thermodynamic study lead to the current, more in depth, work. Concerns not addressed in the earlier studies were questions of how the compressor would perform as the air was progressively cooled as it passed through the compressor. What would be the effect on the stall margin, and how would the engine equilibrium running line be affected by the change in compressor operating characteristics? Additionally, these earlier simulations assumed that the water evaporation in the compressed air would proceed to equilibrium instantaneously. A more realistic model that would take into account the finite rate of heat and mass diffusion through the boundary layers at the water-gas interface was also desired. The current work attempts to address these issues.

COMPRESSOR MODEL

In order to bracket the actual evaporation rate, two models need to be considered. First, the greatest rate of evaporation would occur if the air and water were to reach thermal equilibrium instantaneously. That is, the evaporation of the water would occur in such a manner that the air and water, both liquid and vapor phases, were always at the same temperature. The second model, the results of which are reported here, was used to estimate the lowest rate of evaporation. That is, the evaporation rate was assumed to be diffusion controlled. A number of transient and quasi-steady models have been proposed to estimate the rate of diffusion-controlled evaporation of liquid droplets in a gas (Kent, 1973, Law, 1975, Hubbard, G.L., Denny, V.E., Mills, A.F.,1975, Mills, A.F.,1995.) For the present work, the diffusion-controlled evaporation model was developed following that described by Mills (1995.) In the current model it was assumed that each droplet of water would be entrained in the air stream and travel at the air velocity, thereby minimizing convective effects in the heat and mass transfer. Additionally, all radiation heat transfer effects were neglected.

The rate of liquid evaporation was assumed to follow Fick's mass diffusion law as shown in Eq. (1).

$$(1)\ j_{1,s} A = g_{m1}\ (m_{1,s} - m_{1,e})\, A$$

The mass transfer conductance was estimated by assuming that the Sherwood Number approached it's lower limit

$$(2)\quad Sh = \frac{g_{m1} D}{\rho \delta_{1,2}} \approx 2$$

Substituting Eq. (2) into Eq. (1) yields:

$$(3)\quad j_{1,s} A = \frac{2 \rho \delta_{1,2} A}{D} (m_{1,s} - m_{1,e})$$

In order to estimate the time rate of change of droplet mass due to evaporation

$$(4)\quad \frac{d}{dt}\left(\frac{\pi D^3 \rho_l}{6}\right) = -j_{1,s} A$$

Substituting from above

$$(5)\quad \frac{dD}{dt} = -\frac{4 \rho \delta_{1,2} (m_{1,s} - m_{1,e})}{D \rho_l}$$

An energy balance on the droplet

$$(6)\quad h_c (T_e - T_s) A = g_{m1} (m_{1,s} - m_{1,e}) h_{fg} A$$

The Stanton Numbers for heat and mass transfer were approximated using the Chilton-Colburn Analogy.

$$(7)\quad St = C\mathrm{Re}^{-0.5} \mathrm{Pr}^{-\frac{2}{3}} = \frac{h_c}{\rho C_p V}$$

$$(8)\quad St_m = C\mathrm{Re}^{-0.5} Sc^{-\frac{2}{3}} = \frac{g_{ml}}{\rho V}$$

Combining yields

$$(9)\quad \frac{h_c / C_p}{g_{m1}} = \left(\frac{\mathrm{Pr}}{Sc}\right)^{-\frac{2}{3}}$$

Substituting from above yields Eq. (10) which can be solved for the droplet surface temperature, T_S.

$$(10)\quad C_p \left(\frac{\mathrm{Pr}}{Sc_{12}}\right)^{-\frac{2}{3}} (T_e - T_s) = h_{fg} (m_{1,s} - m_{1,e})$$

The droplet diameter used for the current study was 10 microns. This droplet size insured complete evaporation of the liquid water as it passed through the compressor. Experimental studies have shown that droplet sizes of the order of 10 microns are small enough to insure that the water droplets closely follow the air flow and thereby minimize blade erosion. The compressor model allows for droplets of various size to be used to determine the effect of droplet size on evaporation rate and subsequent effect on compressor performance. A parametric study of performance based on droplet size is planned for the near future.

The compressor design was accomplished by using NACA turning angle and total loss cascade data (Herrig, et.al., 1957) at the mean radius for each stage and then stacking the stages. Each stage was designed as a 50% degree of reaction stage, with a solidity ratio of one, designed to achieve 1/16th of the total enthalpy rise required of the compressor. Flow areas for each succeeding stage were reduced in order to maintain a constant axial velocity during on-design operation. An on-design stage efficiency was calculated to provide an overall compressor isentropic efficiency of 85%. Off-design efficiencies were modeled so as to decrease with off design rotor angles of attack. Positive stall was assumed to occur when the total pressure loss of the rotor reached a value equal to twice the minimum total pressure loss. Stages 1 through 8 were developed using NACA 65-(12)10 blade profiles and stages 9 through 16 were NACA 65-810 profiles.

Isentropic work of a stage was calculated by assuming that the each stage was frictionless and adiabatic and therefore, all of the energy required to evaporate the water came from the cooling of the air. The inlet and exit entropy for each stage was calculated to be the sum of the entropies of the air, the water vapor, and the liquid water at the stage inlet and exit conditions, respectively. The increase in entropy due to the irreversible mixing of the air and the new vapor formed was not included in the calculation. since that increase in entropy would be small. The actual work was then calculated using the isentropic work and the stage isentropic efficiency determined from the stage model.

RESULTS FROM COMPRESSOR MODEL

The compressor model was first run with no water injection. This "dry" compressor map was used to provide the baseline performance with which subsequent simulations with water injection were compared. Figure 5 shows compressor maps for the dry compressor and for 10 gpm water injection rate. As seen in the figure, when operating at the same speed and air flow rates (design air flow rate is 137 lb_m/sec,) the compressor achieves a larger pressure ratio with water injection than with dry operation. The compressor with water injection, also, tends to stall at higher air flows than the dry compressor operating at the same speed. Figure 6 shows the reduction in compressor discharge temperature resulting from the evaporative cooling. This reduced compressor discharge temperature results in a reduced compressor work and would suggest the benefit of increased efficiency in a recuperated engine.

ENGINE MODEL

In order to determine how engine performance is affected by changes in compressor characteristics, as a result of water injection, a simplified engine model was developed. The engine model assumed that the turbine inlet temperature was limited to 2200 F, the combustor pressure loss was 6% of the combustor inlet pressure, and the free power turbine behaved as a simple choked turbine. The performance characteristic maps for the gas-generator turbine were developed using a NASA program (Converse and Griffin, 1984.) The new compressor maps, developed using the evaporative compressor model for various water injection rates were used in the engine model by matching compressor rotating speed, mass flow rate, pressure ratio, and work to the turbine sections. Figure 7 shows engine equilibrium running lines that were developed for the compressor maps of Figure 5. It should be noted, here, that the engine with 10 gpm flow has been fitted with a re-matched free power turbine to better utilize the modified compressor performance. This modification is discussed in the next paragraph.

Engine power curves were developed for the dry case and for water injection rates of 5 and 10 gpm. Figure 8 shows the effect on engine power due to the injection of 5 gpm of water as compared to the dry engine. At each air flow rate the turbine inlet temperature is reduced due do to the water injection and resultant reduction in compressor work. If the fuel is increased to bring turbine inlet temperature back up to near maximum, the engine power is increased by nearly 6% with a subsequent increase in air flow of nearly 5%. Figure 9 shows a similar simulation with 10 gpm water injection rate. For this case a re-matched free power turbine was used to better utilize the additional pressure drop made available by the reduction in work done by the gas-generator turbine. The reduction in compressor work shows up as an increase in free-power-turbine output. With a 10 gpm water injection rate, a power increase of nearly 10% can be obtained with no additional air flow rate. These conclusions are in general agreement with the test results of Stambler (1997.)

Figures 10 shows the engine thermal efficiencies over the power range for the dry engine and for the engine with 10 gpm water injection rate. As shown in the figure the engine power increase is achieved without the inherent loss in efficiency associated with traditional intercooling or injection methods.

SUMMARY

The work presented here has demonstrated that, in addition to increased engine output power, there are several additional advantages of incorporating water-fog injection into the gas turbine engine cycle. These advantages include reduced NOx emissions. Reduced NOx emissions are achieved without the loss in thermal efficiency associated with injecting water into the combustor for NOx reduction. The reduction in compressor power results in lower compressor discharge temperature, again without the decreased thermal efficiency associated with intercooling. The intercooling effect of WFI can be achieved without the complications of multi-spool engines and external heat exchangers. One major advantage of WFI engines for naval applications is the decrease in air flow rate required for a given power (when employing a re-matched free power turbine), which implies a higher specific power. This reduced air flow requirement will reduce the amount of ductwork and the subsequent impact on ship superstructure.

ACKNOWLEDGMENTS

The work described herein was supported in part by Dr. Bruce Douglas of the In-House Laboratory Independent Research Program (PE 601152N) and by Dr. Robert W. Holst of the Strategic Environmental Research and Development Program under it's compliance Pillar (CP-042.)

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Figure 1. Engine NO_X emissions vs. engine power.

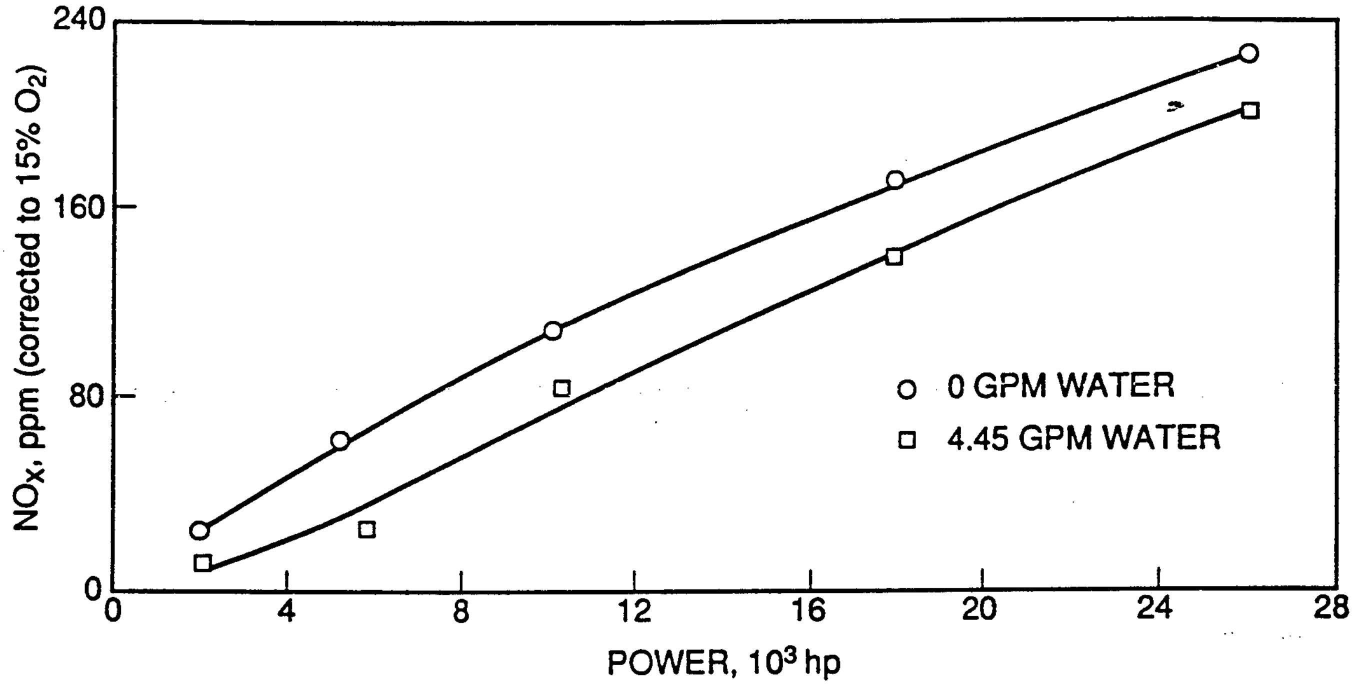

Figure 2. Predicted power increase and specific fuel consumption reduction with water from thermodynamic study.

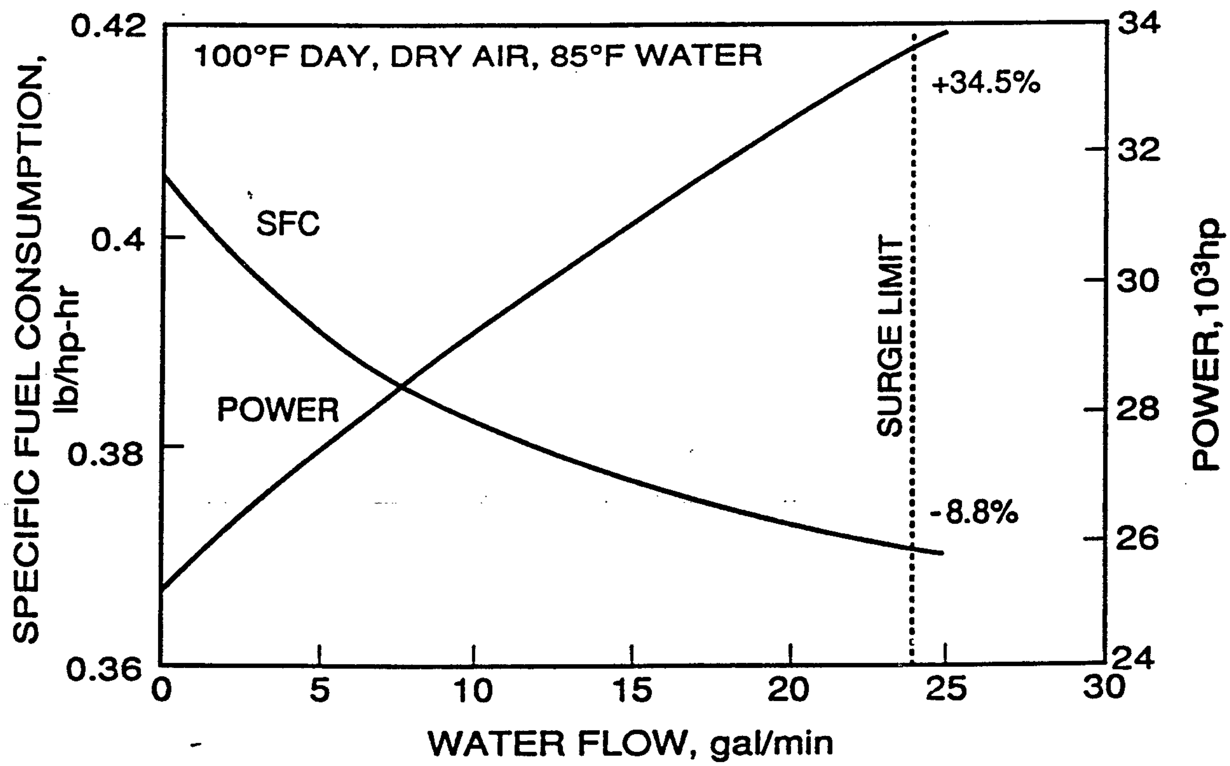

Figure 3. Compressor discharge temperature vs. engine power.

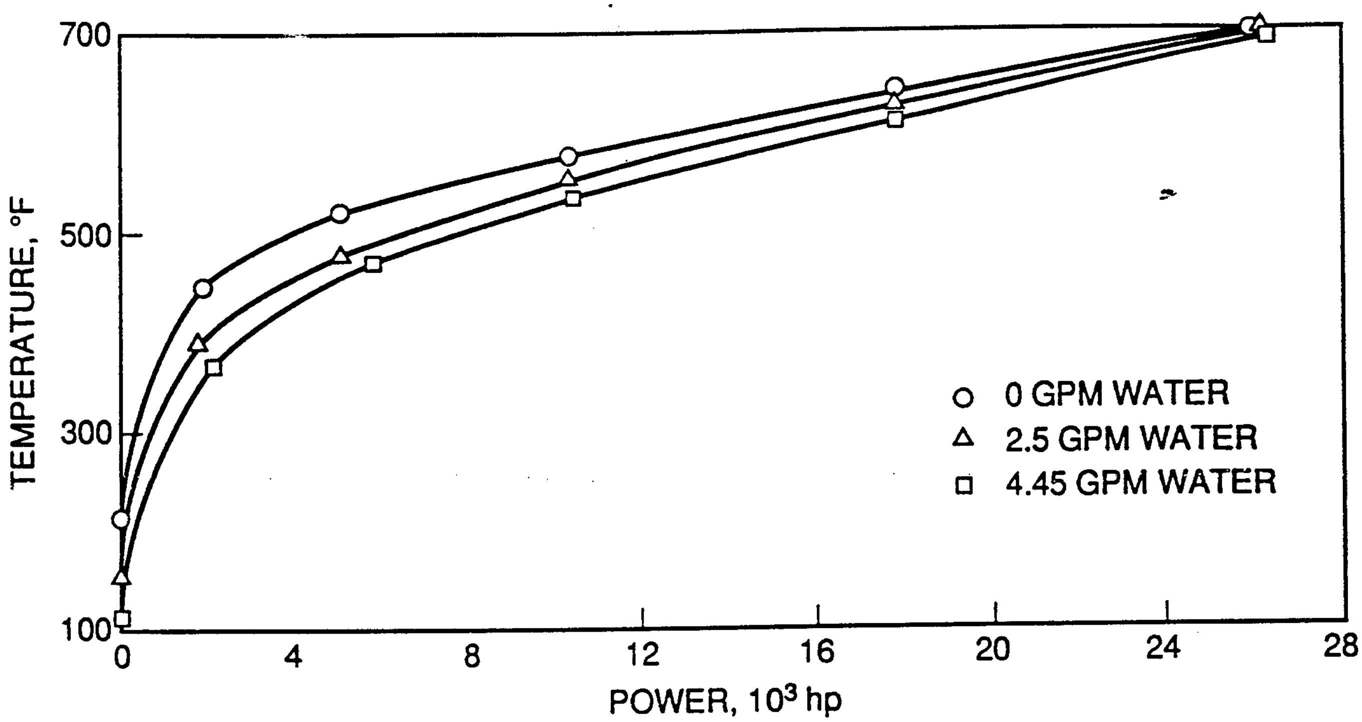

Figure 4. Comparison of recuperated cycles with water-fog injection.

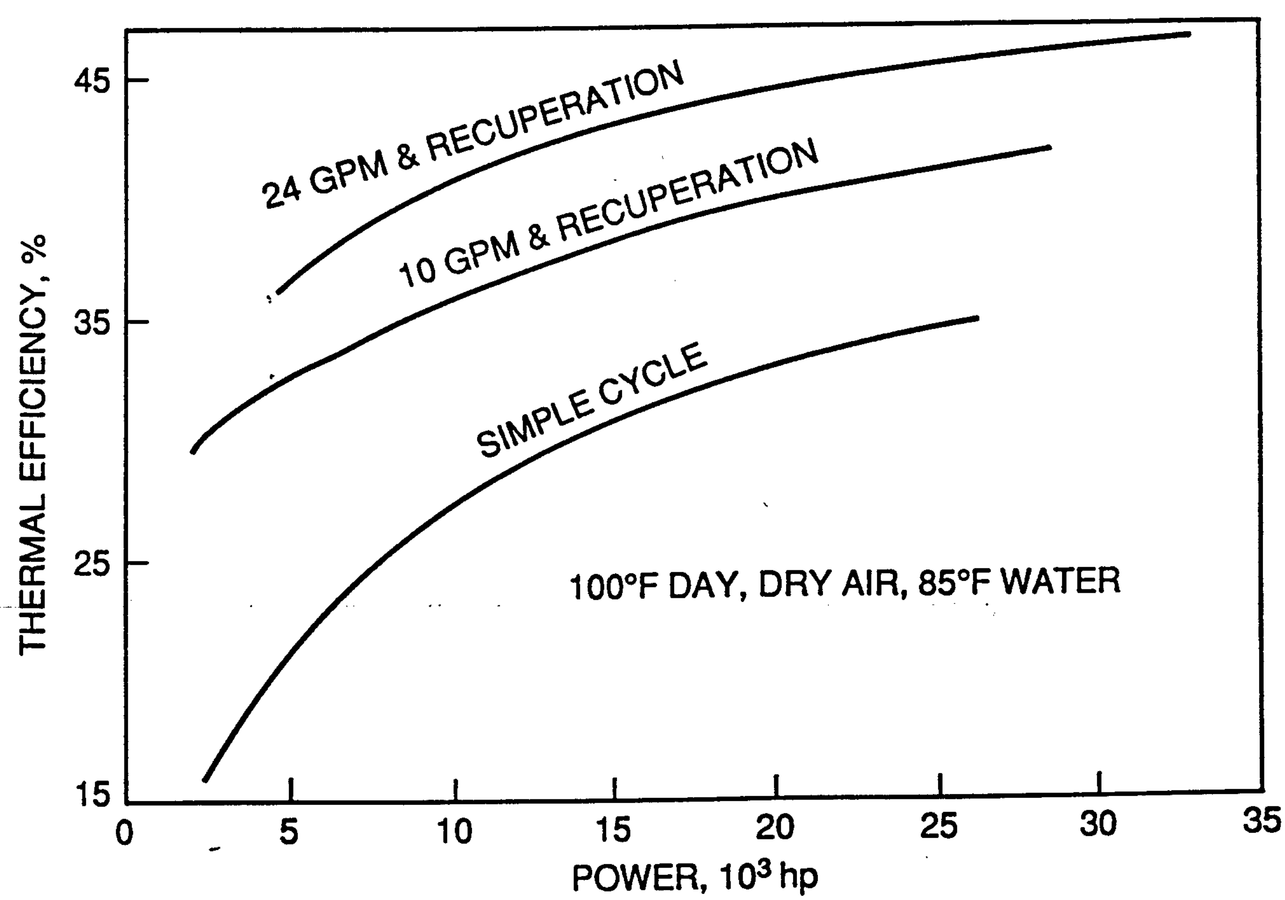

Figure 5. Effect of water flow on the compressor map.

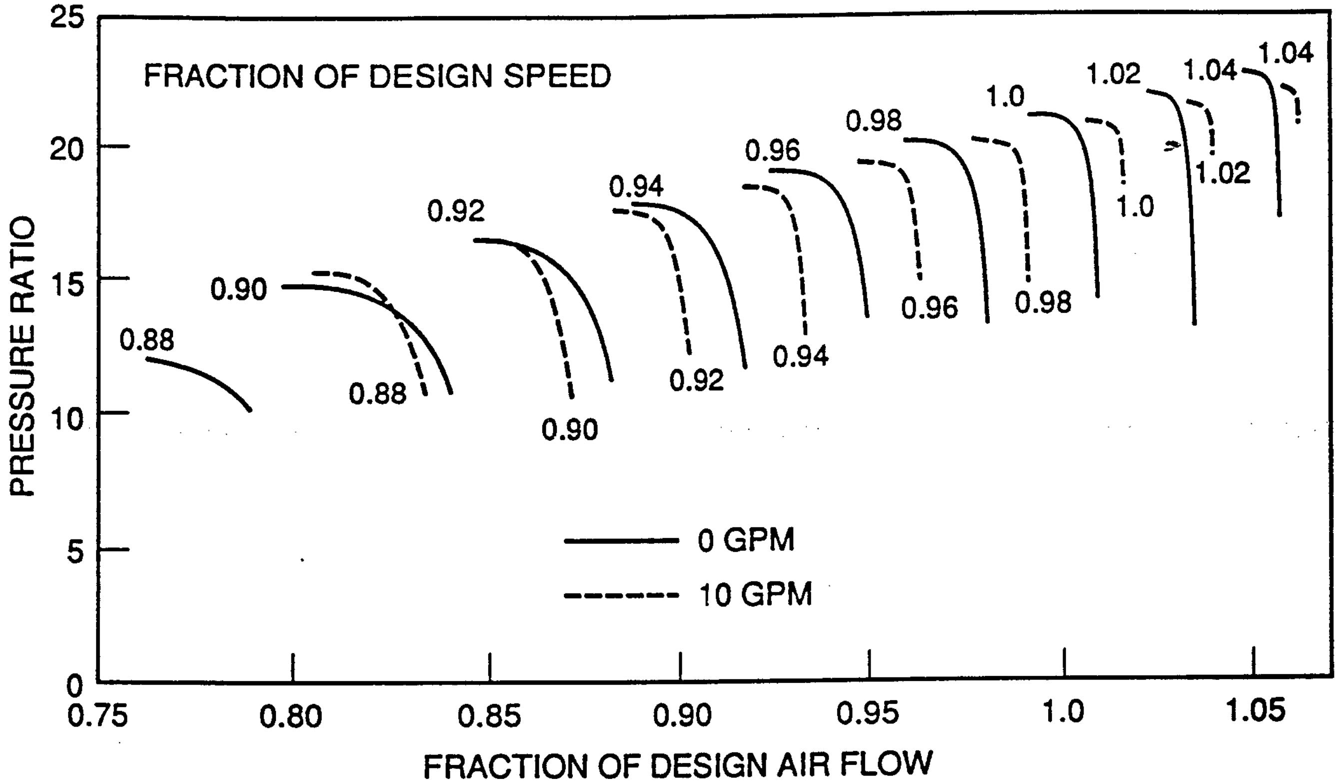

Figure 6. Effect of water flow on compressor-discharge temperature.

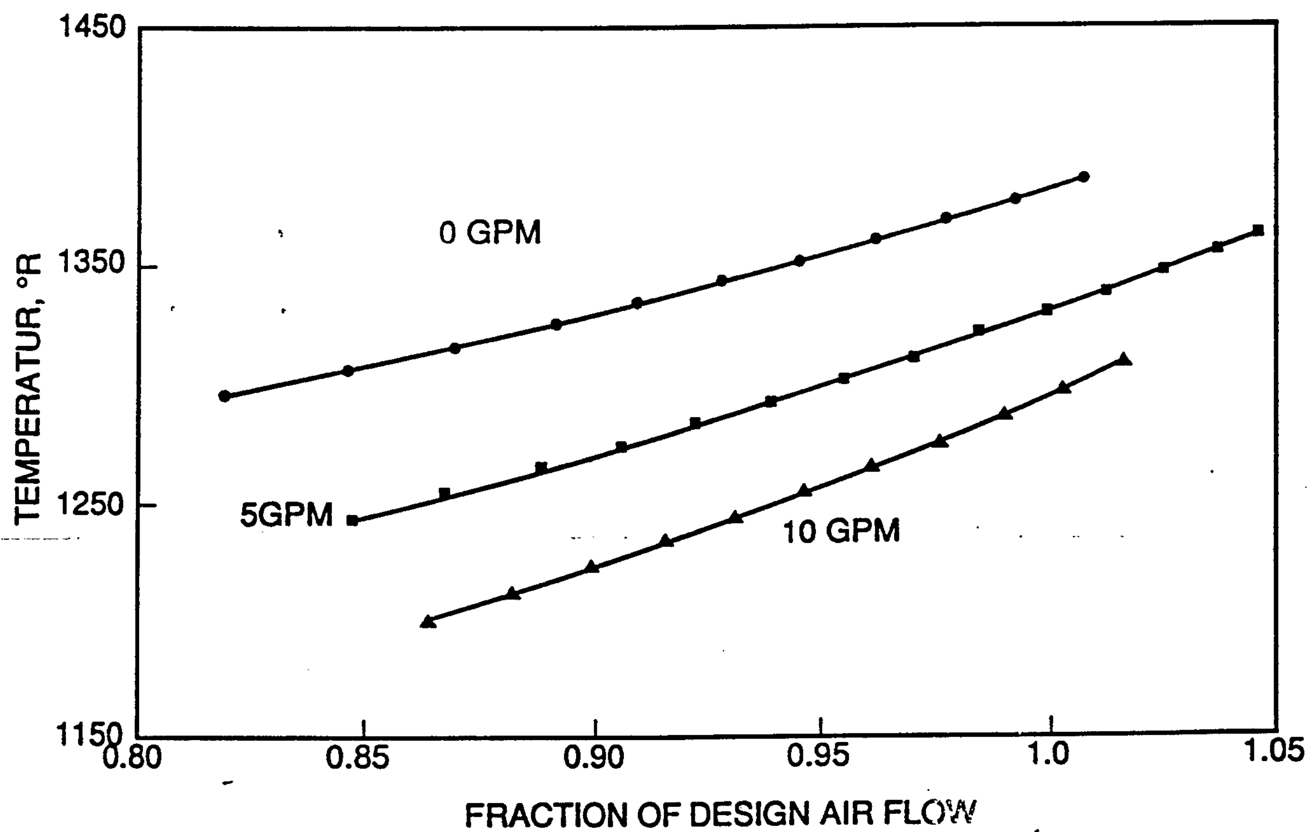

Figure 7. Effect of water flow on the equilibrium running line of the compressor map.

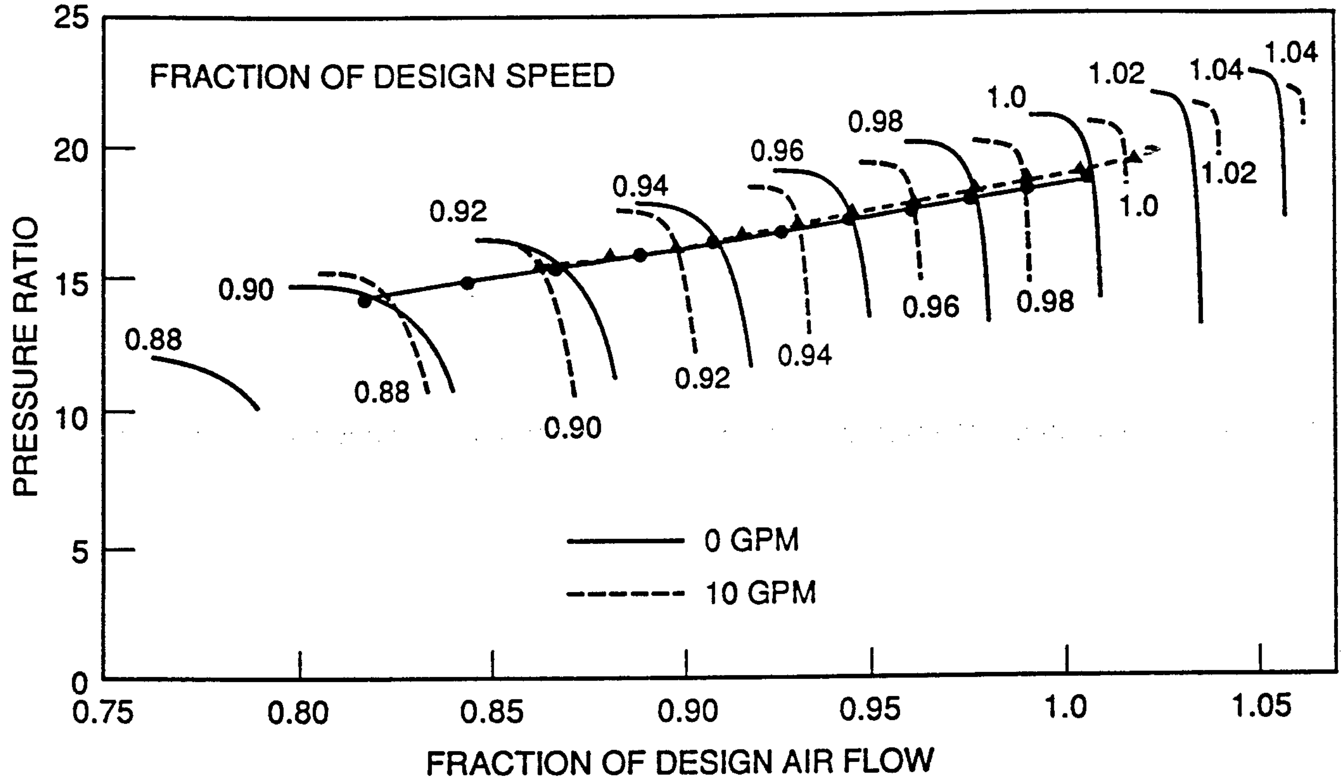

Figure 8. Engine power with 5 gpm water injection compared to dry engine.

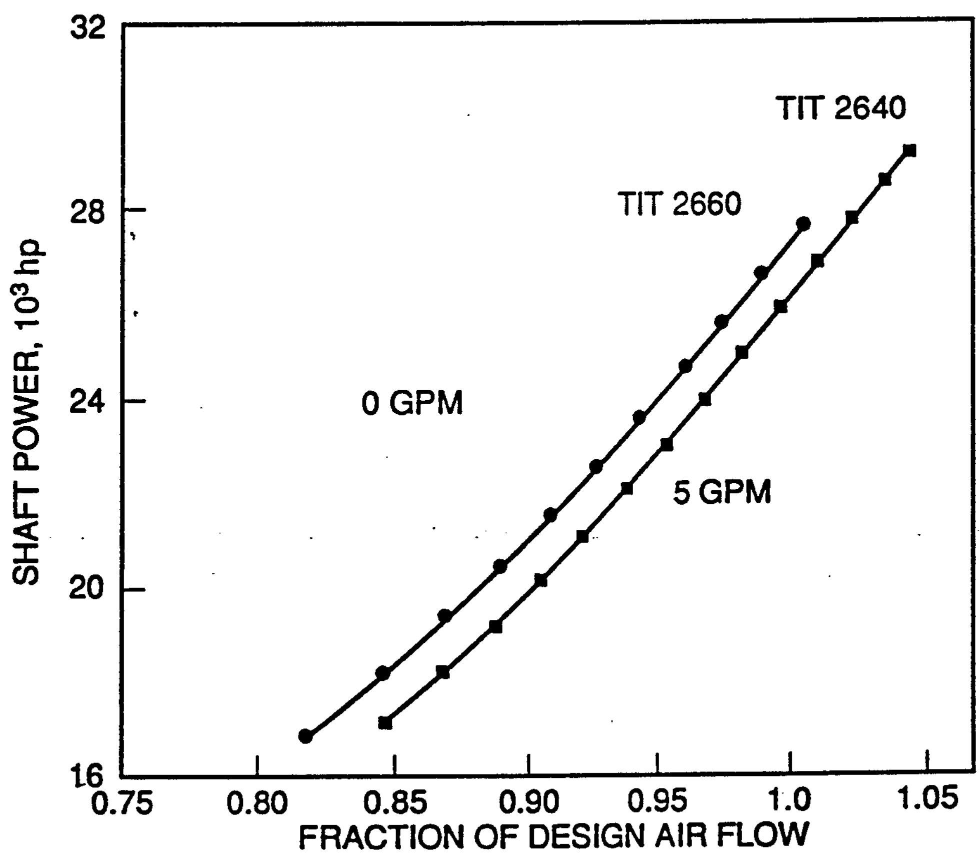

Figure 9. Engine power with 10 gpm water injection and rematched free power turbine compared to dry engine.

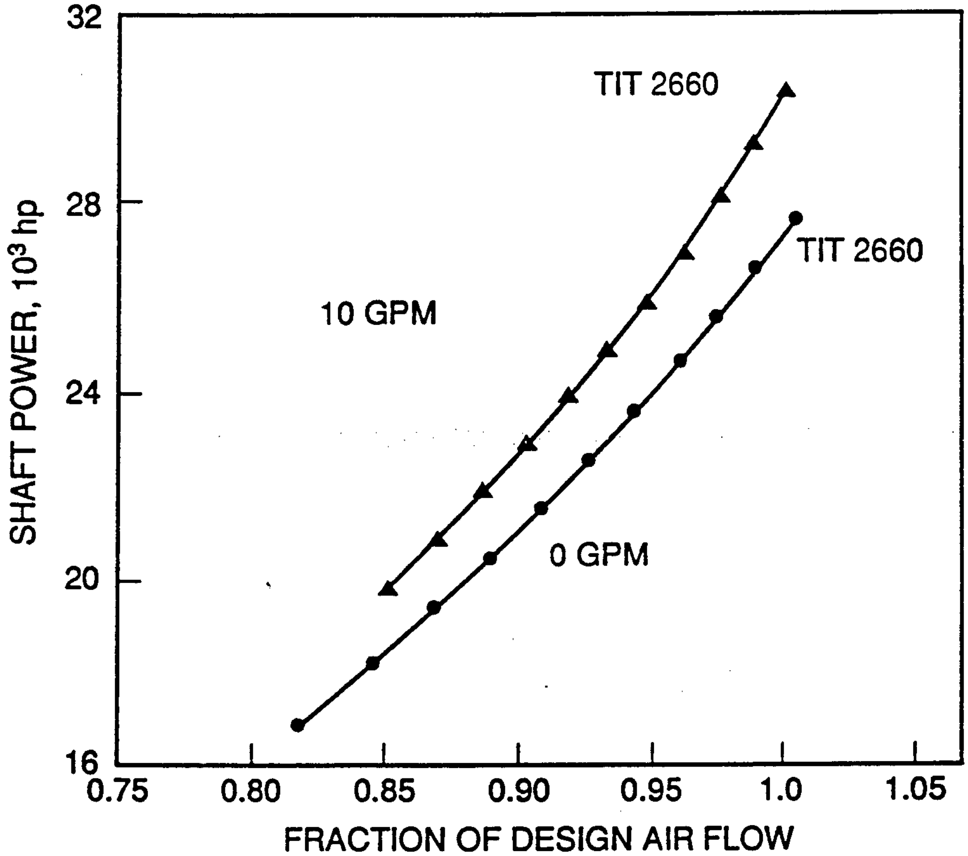

Figure 10. Engine efficiency with 10 gpm water injection and rematched free power turbine compared to dry engine.